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High Efficient C02 Trans-Critical Reciprocating Compressor Part II: Valve Plate and Cylinder Head 3 D Thermal & Flow Optimization

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High Efficient C02 trans-critical Reciprocating Compressor
Part II: Valve plate and cylinder head 3 D thermal & flow optimization

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ABSTRACT

This article is the second optimization step of a semi-hermetic CO2 trans-critical compressor based on a fully 3D multi-physics compression process simulation. The model will be detailed and its experimental validation phase will be discussed.
A complete results analysis will be carried out based on pressure pulsations, valves dynamics, thermal exchanges, cylinder filling, inlet and discharge lines influences.
Finally, simulated improvements will be validated by experimental testing of a prototype compressor.

1. INTRODUCTION

As explained by ASERCOM statement about Carbon Dioxide in RAC Systems, CO2 was one of the first refrigerants to replace early air cycle systems and was in use primarily for shipboard refrigeration in the beginning of the twentieth century. It was then superseded by chlorofluorocarbons. However, since CO2 is environmentally benign, non-toxic (in the classical sense), non-flammable, chemically inactive and offers a very high volumetric cooling capacity together with excellent heat transfer properties, it is increasingly considered for use today in RAC systems. Because of its very low global warming potential (GWP=1) and zero ODP, CO2 systems do not need the very stringent containment criteria necessary for HFCs and other refrigerants. Since CO2 is in the same safety class (L1) as HFCs the safety requirements may be less onerous than they would be for ammonia or hydrocarbons.
Thermodynamic characteristics of CO2 are much different to refrigerants usually applied in RAC system. This is mainly related to the very low critical temperature of 31°C which, depending on the heat sink temperature on the discharge side, may require so-called trans-critical operation. The system design for trans-critical operation will differ from a conventional vapor compression cycle.
The energy efficiency tends to be lower as compared to a sub-critical conventional system. Nevertheless, in several applications or in specific circumstances CO2 systems can reach or exceed the energy efficiency of systems with established refrigerants. In any case, a high efficient CO2 compressor is required to be competitive versus HFC solutions in terms of TEWI.

2. COMPRESSOR SPECIFICATION

The compressor is developed for the following applications:

1. Water Heat Pump for Sanitary Application or Heating mode.
2. Trans-critical Refrigeration.

And for the following size:

Mass flow 850 kg/h with suction pressure 34.9 bar (evaporating temperature 0°C) and discharge pressure 95 bar.

3. COMPRESSOR DESIGN

To reach high efficiency with a CO2 trans-critical semi-hermetic reciprocating compressor, we must find the good compressor architecture; that is to say, the best compromise between the various losses:

- leakage losses (=> high speed, small bore/stroke ratio, limited number of cylinders);
- heat transfer losses;
- friction losses (bearing, piston, piston rings);
- flow losses (=> low speed, large cross section, large volume, multiple cylinders).

Of course reliability criteria is also considered (limitation of bearings load and speed, consideration about oil management & motor cooling…).

Based on an analytical compressor simulation tool and on DCC know-how, the following compressor characteristics have been selected:

- High speed compressor. That is to say, the compressor is directly driven by a 2 poles AC motor (2900 rpm at 50Hz / 3500 rpm at 60Hz). Whereas, typically, HFC semi-hermetic reciprocating compressor of similar size runs at 1450 rpm.
- Vertical crankshaft.
- Single crank crankshaft and opposed cylinders.
- Bore/stroke ratio ~ 1.
- 2 pistons.
- 2 piston rings.
- Annular inlet and discharge valves (see figure 5 and reference [1]).

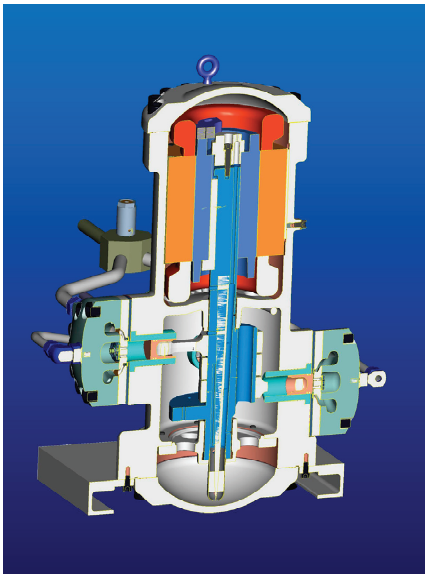


Figure 1: Compressor axial cutaway view (Prototype I)

4. COMPRESSOR 3D STEADY STATE MODEL

A first prototype (called prototype I) was designed, manufactured and tested. To reach better efficiency an optimization work on compression process was launched. Thanks to CFD analysis, the main local pressure losses were identified. The re-design work then focused on reducing the losses in a second optimized prototype (called prototype II).

4.1 Stationary CFD Modeling

First round of 3D simulation was performed by 3D CFD stationary analysis.
Two fluid domains were defined (see figure 2) for suction and discharge path.

- Inputs / boundary conditions:
  Suction (respectively discharge) nominal static pressure; suction nominal temperature; discharge measured temperature; mass flow (we used average mass flow during respectively inlet and discharge process); gas density.
- Valves : 100% open.
- Gas : incompressible (gas density is an input).

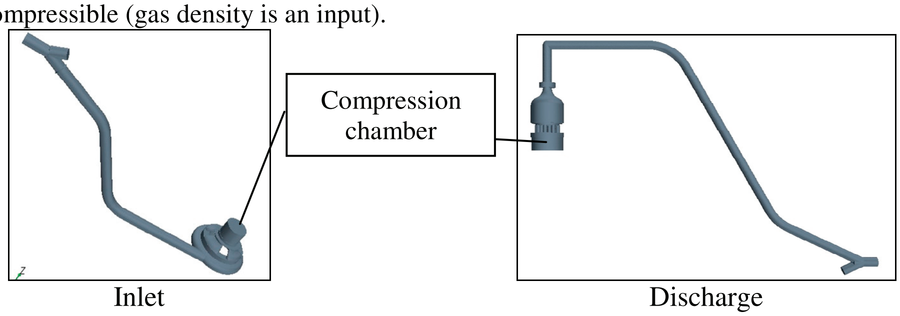


Figure 2: Compressor Prototype I - Fluid Domains

4.2 Compressor Re-Design

Based on the first batch of domains calculations, inlet and discharge line (cylinder heads and piping) were modified. Valve plates geometries were also improved: flow losses reduction led to greater valve lift and lower valve stiffness which gave additional efficiency gains. Main changes were:

- **Compressor inlet line.**
  A new inlet pipe position was selected (from tangential inlet to radial inlet) to minimize flow recirculation inside suction plenum (see figures 3 and 4). Moreover effective plenum volume was increased by additional specific feature.
- **Valve plate - suction**
  New valve lift (+25%) & reduced valve stiffness (-75%). Obviously, if flow section is increased, flow velocity and thus losses are reduced. An aerodynamic device (45° chamfer) was defined for the transition between valve plate suction axi-symmetrical recess and valve plate suction port (see figure 5).
- **Valve plate - discharge**
  New valve lift (+25%) & reduced valve stiffness (-50%). New valve plate discharge flow paths: from 8 cylindrical tubes to 4 slots with smoother aerodynamic transition with axi-symmetrical discharge valve recess (see figure 5).The main goal of this device was to reach homogenous speed (axi-symmetrical speed profile) across discharge valve port (see results in figure 6).
  Several 45° chamfers were added: one at slots inlet and one at discharge recess outlet.
- **Discharge line.**
  Compressor architecture requires a 90° gas turn at cylinder head outlet. In prototype I, this 90° gas turn was made by a simple 90° elbow resulting in high fluid speed and large recirculation at elbow outlet (see figure 7). The idea was to make this gas turn while gas velocity is constantly and slowly increased (section is reduced) to avoid the creation of any flow recirculation area. Moreover plenum volume was increased.
  - Cylinder head
    In order to reduce heat transfer from discharge to suction plenum of cylinder head, an air separation was created between these plenums (see figure 8).

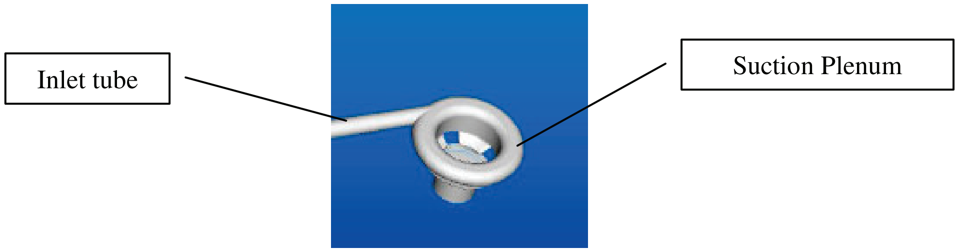
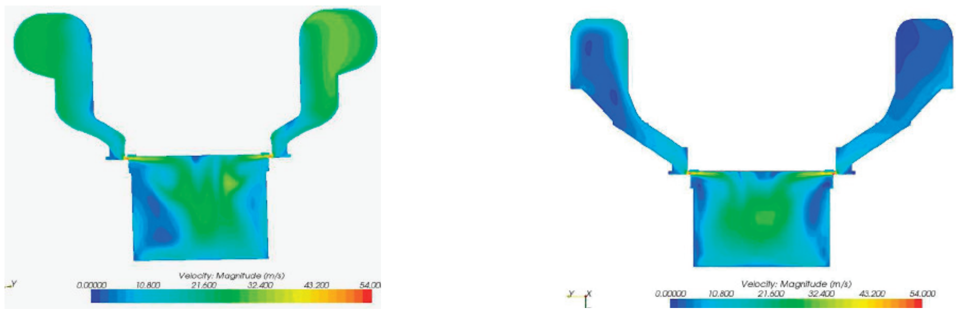


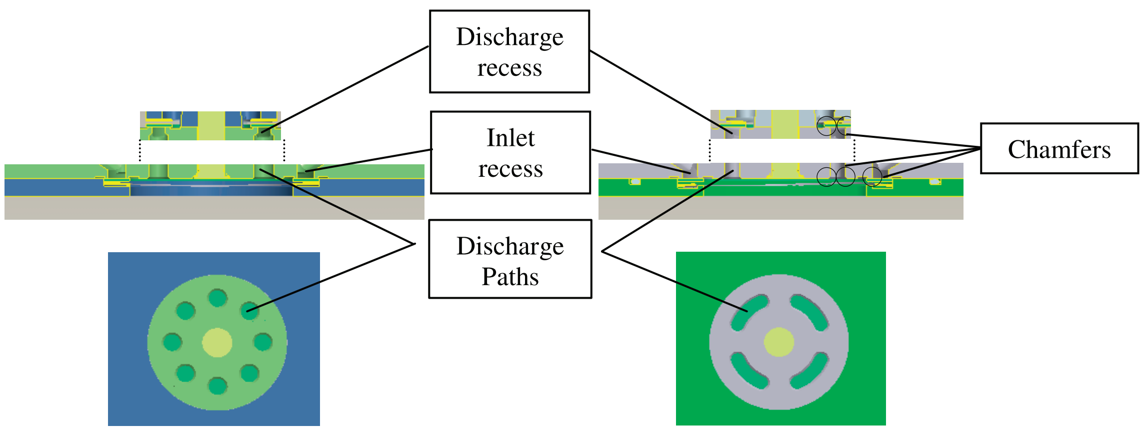
Figure 3 Prototype I with its tangential inlet tube



Prototype I
Tangential inlet tube

Prototype II
Radial inlet tube

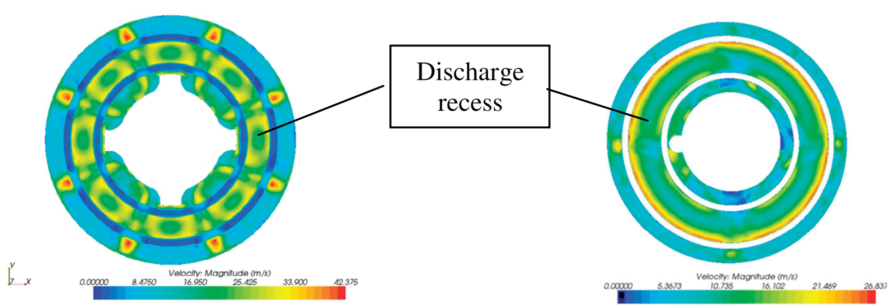
Figure 4: Velocity magnitude inside Compressor suction plenum and compression chamber (suction phase)



Prototype I

Prototype II

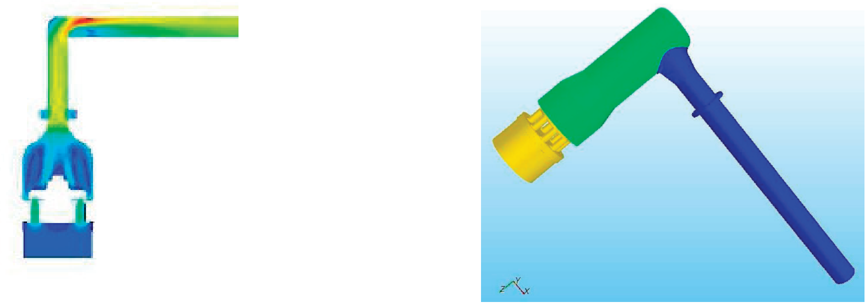
Figure 5: Valve plates



Prototype I

Prototype II

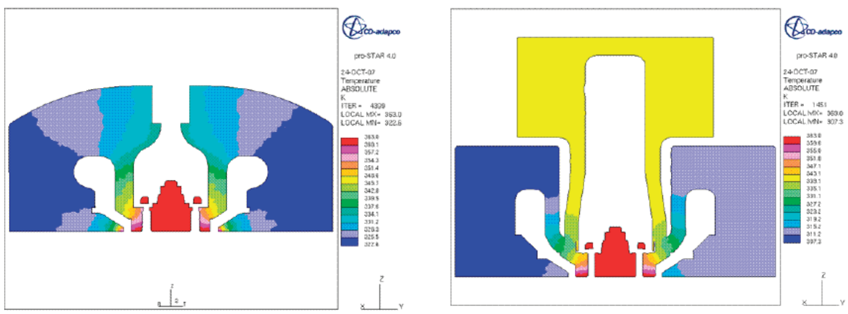
Figure 6: Velocity magnitude across discharge recess



Prototype I
Velocity magnitude

Prototype II
Fluid Line

Figure 7: Compressor discharge lines



Prototype I

Prototype II

Figure 8: Thermal analysis of cylinder heads

4.3 Compressor Re-Design results

All the improvements detailed in § 3.2 were applied to prototype II. This prototype was manufactured and tested leading with a 20% global isentropic efficiency improvement versus prototype I (see table 1).

			Unit	**Prototype I**	**Prototype II**
Suction line	Compressor	Pressure Losses	bar	0.36	0.18
			Adim.	100	50
	Valve Plate	Pressure Losses	bar	1.49	1.03
			Adim.	100	69
Discharge line	Valve Plate	Pressure Losses	bar	2.15	0.90
			Adim	100	42
	Compressor	Pressure Losses	bar	1.72	0.04
			Adim.	100	2
Cylinder head		Heat transfer Flux	Adim.	100	85
Semi-Hermetic Compressor		Volumetric Efficiency	Adim.	100	130
		Global Efficiency	Adim.	100	120

Table 1: summary of design improvements and efficiencies of Prototypes I & II

5. COMPRESSOR 3D UNSTEADY MODEL

5.1 Unsteady CFD Modeling

The reciprocating compressor process is very complex and strongly unsteady. As a consequence, analytical tools and steady state 3D approaches are limited to perform numerical optimization loops and reach optimal efficiency.
On the other hand, compressor testing is very time and cost consuming, so a limited number of data are often measured as:

- cooling capacity
- input power
- temperature at few locations
- pressure at few locations

Consequently, local optimization can only be done through a "virtual" prototyping.
The aim of this paragraph is to detail the main features of the 3D unsteady tool that DCC is being developing for the final step of CO2 compressor optimization.

Model description

- Inputs :
    a. Inlet gas conditions: average static pressure and temperature.
    b. Average static discharge pressure.
    c. Piston movement law : table (time; axial piston position).
    d. Fluid thermodynamic properties.
    e. Solid thermal properties.
- Moving fluid mesh in order to follow piston and valves movements.
- The gas is considered as compressible according to real gas properties (resolution of a real gas equation of state).
- Multi-physic : coupling between flow and thermal:
    a. Current tool : weak coupling (first steady state analysis to compute solid thermal field and unsteady muti-physic analysis for gas compression process);
    b. Nest step: strong coupling (full transient analysis).
- Multi-physic: coupling between flow and valves mechanics. For simplicity, valves motion law is only mono-dimensional:

$$m.\ddot{x} = Fs(x) + Ff$$

With
m: equivalent valve moving mass
x: valve position
Fs(x): spring load on valves (see figure 9)
Ff: resultant of pressure load on the valve (computed by CFD software)

That is to say, in numerical form
v(n) = Ff(n-1)*dt/m + Fs[x(n-1)]*dt/m + v(n-1) and x(n) = v(n)*dt + x(n-1)

n, n-1: time iteration number
dt: time step
v: valve speed

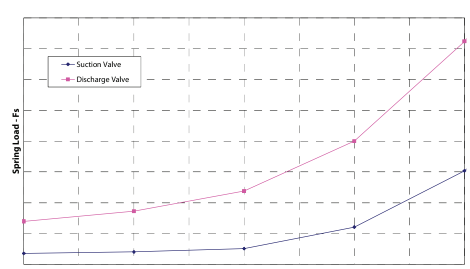
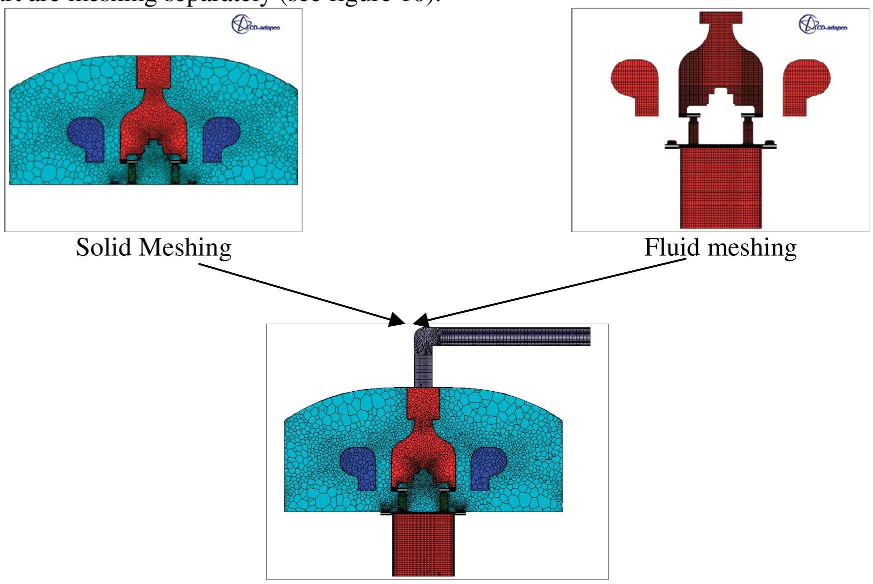


Figure 9: spring load as a non linear function of valve position

Meshing

Fluid and solid part are meshing separately (see figure 10).



Final meshing for unsteady multi-physic (thermal+flow) CFD simulation

Figure 10: Principle of CFD meshing

5.2 Results & Model Validation (with prototype I)

The simulation of prototype I design required one week of CPU time. Results (figure 12) were compared to measured data (see figure 11). The model is able to predict properly enough fluid dynamic to be used as design tool to make relative comparison between two designs:

- valves movements are realistic;
- pressure fluctuations are in the right order of magnitude.

Nevertheless, an improvement of boundary conditions (strong influence of cylinder#2 on cylinder#1 initially neglected for the model) and fine tuning should be done to compensate simplifications in the model (rebound law, oil sticking influence, valves deformation, leakages …).

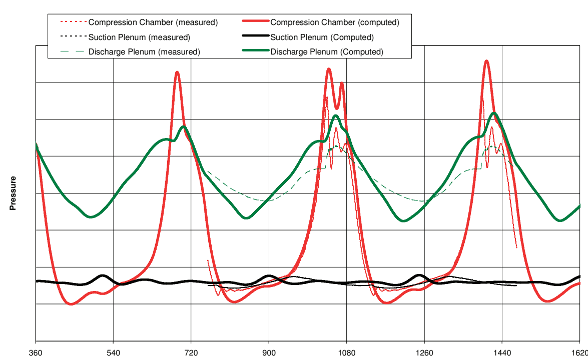


Figure 11: Comparison of experimental and simulated pressure

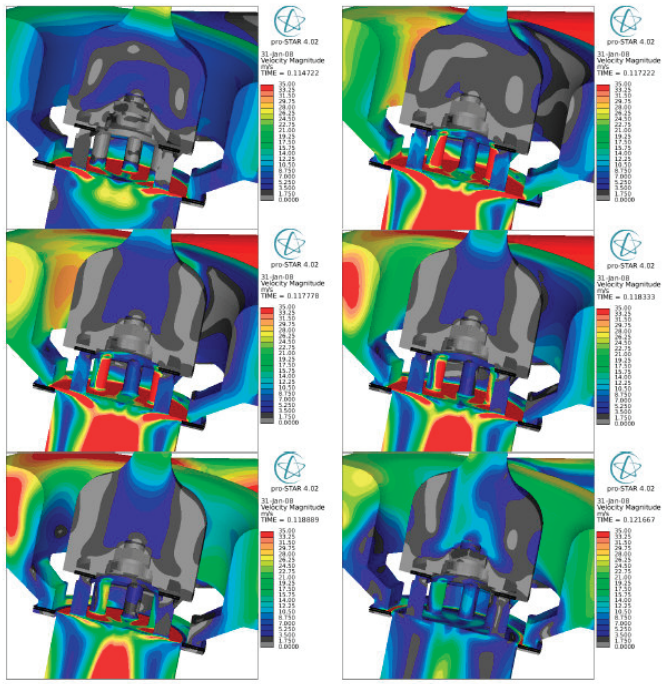


Figure 12: flow velocity magnitude at various time during suction process

6. CONCLUSIONS

In this article we described a new high efficient CO2 trans-critical compressor. The utilization of CFD to optimize efficiency was detailed. First round of improvement was performed based one simple steady analysis, and experimental PV-diagram (see reference [1]), leading to the definition of a second prototype. This compressor prototype is already meeting the highest market level efficiency. An unsteady multi-physic 3D tool was developed in parallel and validated versus experimental measurement. Another round of prototype design optimization will be made thanks to this dedicated multi-physics tool to make an additional step of energy efficiency improvement.

NOMENCLATURE

ASERCOM	Association of European Refrigeration Compressor and Controls Manufacturers.
RAC	Refrigeration and Air-Conditioning Systems
ODP	Ozone Depletion Potential
GWP	Global Warming Potential
TEWI	Total Equivalent Warming Impact
HFC	Hydro-Fluoro-Carbon
DCC	Danfoss Commercial Compressors
CFD	Computational Fluid Dynamics
CPU	Central Processing Unit of a computer

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[1] Dugast P., Bonnefoi P., 2008, *High Efficient CO2 trans-critical Reciprocating Compressor – Part I : Valve plate and cylinder head 1 D flow optimization,* International Compressor Engineering Conference at Purdue

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